Design Solution for a Device to Determine the Energy Consumption of Sawing Wood with Chain Saws

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Cutting wood chips is one of the common work procedures in the course of processing wood with a saw chain. This paper presents the conceptual design of the device and the design of the device for determining the energy demand of such a process. The preparation is designed with the given parameters to achieve the most realistic conditions for sawing wood. It must contain the original cutting device from the STIHL MS 261 chainsaw. It consists of several calculations to determine the cutting moment and the loaded points of the preparation, which define the possible place of damage. In the conceptual design, calculations for cutting force Fr = 924 N and tensile force Fn = 290 N are processed in the work. From the calculated forces, it is possible to determine the resulting stress at the attachment or critical points. For comparison, a finite element method analysis was created in Creo Parametric. The results of the analysis confirm the appropriate selection of material and parameters for the application of the preparation to the test device. The resulting voltages, corresponding to a maximum value of 132 MPa, are negligible for this type of preparation in terms of safety and durability.

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Keywords: Wood cutting; Saw chain; Finite element method; Cutting device

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INTRODUCTION

The development of new technologies, the reduction of energy requirements, and reliability in wood processing are forcing us to increasingly investigate the basic issue of wood sawing with the use of new technologies (Kovac *et al.* 2021, 2022; Landekić 2023, Wójcik 2007). Currently, high demands are placed on forestry and woodworking machines, especially regarding their negative impact from an ecological point of view. Given current requirements, preference is given to machines having suitable ecological and ergonomic parameters. The entire process of wood processing, which begins with felling the tree, depends on the energy demand and speed of individual operations, *i.e.*, filing, branching, shortening, and manipulation. For the mining-production process chain to be functional, it is necessary to optimize each work operation. An important task is to optimize the technical-technological parameters of the cutting mechanisms of the equipment with high requirements for their reliability, low energy consumption, cutting performance, weight, and cut quality. To optimize the cutting mechanisms, it is necessary to test them and investigate the effects when cutting wood. It is also necessary to focus on the negative impact on people. A cutting device with a saw chain is a source of vibration and noise during its operation. The solution to the mentioned requirements may require innovation and adaptation of saw chain mechanisms to new working conditions (Krilek *et al.*2023; Kuvik *et al.* 2017, 2021; Antonić 2023; Maciak 2017). For the functionality of chainsaws, it is possible to provide different types of motor drives, such as electric motors, combustion, pneumatic, or hydraulic motors. The drive of ordinary chainsaws is provided by a combustion two-stroke engine. Electric saws are also applied in practice, and these achieve the necessary performance and excellent ergonomic properties in operation (Poje 2020). In contrast, hydraulic drives are usually mounted, *e.g.*, in harvester heads. Despite significant technical progress and the threats associated with the use of combustion saws, even heavy mobile machines such as cutters and harvesters have not been able to push them out of the market. (Skarżyński and Lipiński 2013). A single-cylinder internal combustion engine is characterized by a variable movement of the piston during the execution of the work cycle. It accelerates when the mixture is ignited in the combustion chamber and slows down during the compression cycle. This results in a discontinuity in the velocity and angular acceleration of the crankshaft (Gendek 2006). This translates into high chainsaw acceleration values, resulting in significant inertial forces that are often several times greater than active cutting forces (Więsik 2007; Maciak 2015, 2017).

One of the main parts of a chainsaw is the cutting device, which includes the saw chain. The cutting elements (teeth) in chainsaws are connected into a single unit. The saw chain moves freely along the guide bar. The guide bar directs the saw chain in the desired direction. It ensures that it remains rigid and sets the direction of the cut. Poor kinematic connections between the system of cutting elements and the body of the cutting member enables a more rational path of movement of the saw chain, which reduces the outline dimensions of the cutting mechanism in relation to the cut surface (Kováč *et al.* 2013; Siklienka *et al.* 2017; Marenče 2017).

Modern technical solutions have improved work safety and ergonomics as well as increased work efficiency of chainsaw operators (Gendek and Oktabiński 2012). Even with progressive changes in chainsaw design, at some stages of sawing, various events can reduce the efficiency of the wood cutting process. The blunting of chain teeth while cutting wood is one of the most important aspects, directly connected with the cutting process. Even the latest solutions in blade geometry have not been able to eliminate this problem. During chainsaw operation, the chain tension can change dynamically. The reason behind this is the non-uniformity of speed and angular acceleration of the crankshaft, which is especially noticeable in single-cylinder engines (Gendek 2006). This translates into high values of the chain acceleration, resulting in high inertia forces. The values of inertia forces are often several times higher than the value of active cutting forces (Więsik 2007).

Experimental Device for Testing Cutting Conditions

The experimental device is shown in Fig. 1. It is protected by the utility model filing “Device for measuring the cutting conditions of tools” utility model number 8298. The experimental device consists of two parts, a cutting part and a sliding part. The drive of the cutting mechanism (3) is provided by a three-phase SIEMENS electric motor (8) with a power of 7.5 kW. The torque is transmitted *via* belt transmission (7) to the shaft housed in the bearing housing (5). The torque then passes through the HBM T20WN torque sensor (6), which is equipped with Giflex GFLL-38 safety couplings (4) at the input and output to prevent damage to the sensor. A specimen holding mechanism (2) is attached to the sliding mechanism (1).

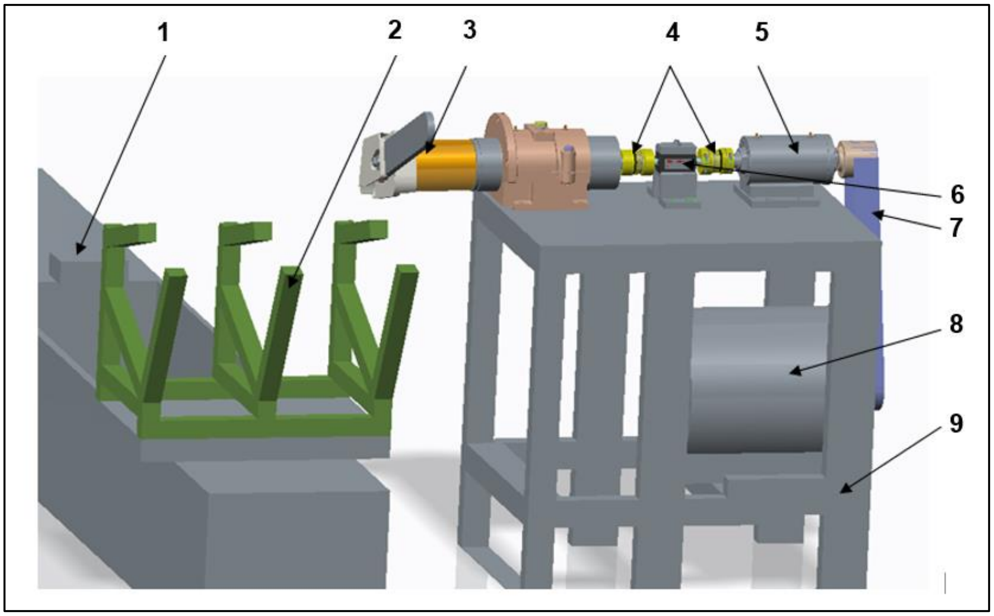

Fig. 1. Model of the experimental equipment

METHODOLOGY

The structural design of the fixture for holding the chainsaw cutting device consists of two main parts, as shown in Fig. 2. The housing (1) contains a shaft on which the engine block (crankcase) of the chainsaw with the cutting device (2) is located. The cutting device is from a STIHL MS 261 chainsaw.

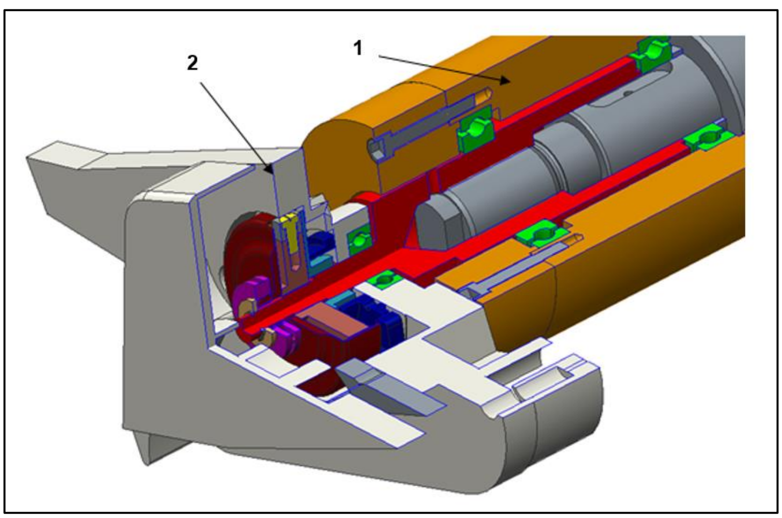

Fig. 2. Model of the fixture for holding the cutting mechanism in the cut

Figure 3 shows a model of the cutting device. The engine block (1) rests on the designed shaft and is fixed to the housing cover with three M8 socket head cap screws (14). The original crankcase bearing, which required lubrication, was replaced with a 6202 bearing (3). Chain lubrication is provided by the oil pump (5) supplied for this type of chainsaw. The drive of the pump is provided by the screw (6), which is driven directly by the clutch drum (10). To transfer the torque to the clutch drum (10), a washer (9) is used, which replaces the original clutch due to the low revs of the stand, at which it would not work. The washer transmits the torque using a pin (8), which is fixed by an M5 screw with a countersunk head (7). A chain ring (11) is located on the clutch drum, which is secured by a washer (12) and a size 10 circlip (13). The entire device is protected by a cover (2).

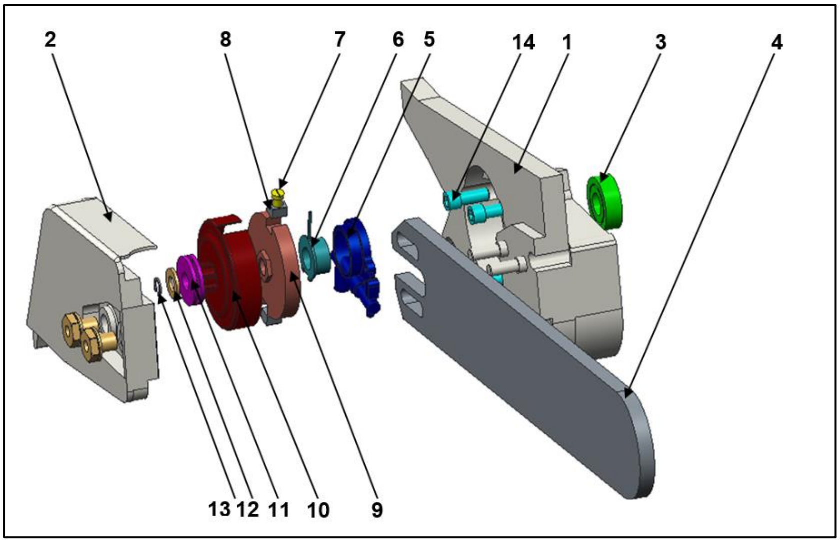

Fig. 3. Cutting Device STIHL MS 261

The model of the bearing unit in the section is shown in Fig. 4. In the bearing unit (1) there is a bearing 6008 (4), which is directly on the shaft of the experimental devices. The bearing is secured by a size 68 circlip (6). The shaft (3) fixes the bearing 6008. The bearing 6010 (5) is located on the shaft (3) secured by the cover (2), and it is fixed to the bearing unit with four M6 screws with a cylindrical head with an internal hexagon (7). Assembly of the cover (2) is possible only after fitting the shaft (3) to the output shaft of the experimental device and securing it with M6 headless screws (8). The limiting ring (9) ensures the distance of the bearing from the mounting on the shaft of the experimental device.

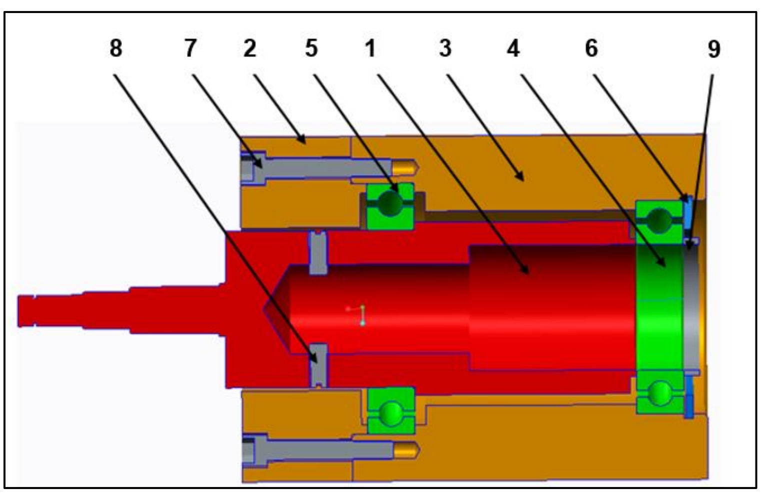

Fig. 4. Bearing Unit Model by Experimental Devices

The shaft (Fig. 5) is made of a ø 55 mm hot-rolled round bar EN 41 523. It is attached to the output shaft of the experimental device using an M 30 x 2 thread. Because the output shaft rotates in the direction in which the designed shaft would loosen on its own, it is provided with two holes with M6 thread for the placement of safety screws. The milled surfaces allow the nut to be tightened using a standard size 41 wrench. The shaft has a surface adjusted to a roughness of 0.8 for bearing 6010. An offset is designed on the largest diameter, as the shaft presses the bearing 6008 located on the stand shaft with this surface. The front part of the saw is based on the dimensions of the original engine block and the saw's cutting mechanism, STIHL MS 261. The transition to ø 15 is provided with a fitting for the crankcase bearing. The surface with a ø 15 mm on a length of 30 mm is adjusted for the storage of the bearing. An M 12 thread is made to hold the washer. The end of the shaft is adjusted to a diameter of ø 10 mm along the threaded surface and there is a groove for the safety ring at the end.

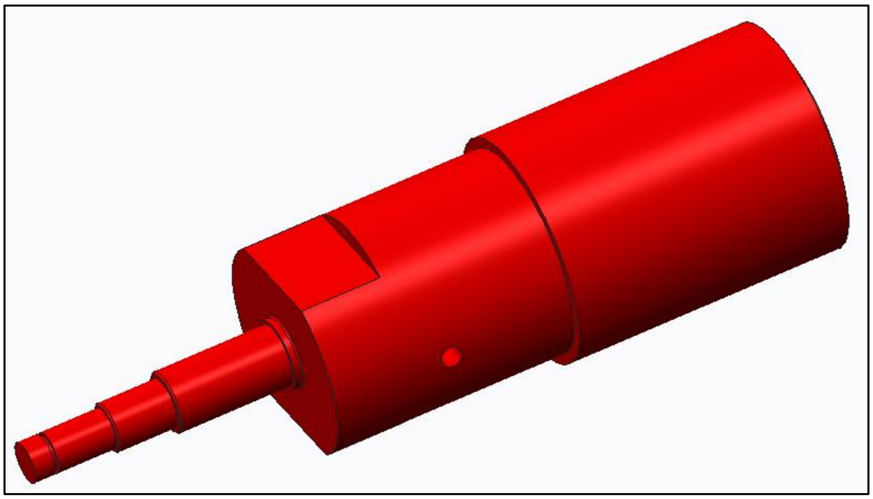

Fig. 5. 3D model of the shaft

The main part of the device is the shaft, which is subjected to torsional stress. Using mathematical calculations, the moment (Eq. 4) on the radius of the drive sprocket d = 20 mm caused by the cutting force F_r was determined. A saw chain with a division of 0.325" = 8.255 mm for a bar of length 37 cm has 31 teeth. The calculations were applied for dry pine. The calculations of cutting forces were determined according to Mikleš (2011). Cutting speed was calculated according to Eq. 1,

$$v_c = \pi \cdot D \cdot n = 3.14 \cdot 0.032 \cdot 3000 = 301.4 \text{ m} \cdot \text{min}^{-1} \tag{1}$$

where D is the diameter of the drive sprocket 32 mm => 0.032 m and n is the revolutions (s^{-1}), 3000 min^{-1}. The feed rate was calculated according to Eq. 2,

$$v_f = \frac{f_z \cdot v_c}{t_h} = \frac{0.1.10^{-3} \cdot 5.024}{16.51 \cdot 10^{-3}} = 0.03\, m \cdot s^{-1} \tag{2}$$

where f_z is the feed per tooth (m), v_c is the cutting speed ($m.s^{-1}$), v_c = 301.44 $m.min^{-1}$ = > 45.024 $m.s^{-1}$, and t_h is the distance between teeth (m). The f_z value was chosen according to Mikleš (2011). The cutting force was calculated using Eq. 3,

$$F_r = k \cdot b \cdot H \cdot \frac{v_f}{v} = 29.3 \cdot 10^6 \cdot 8 \cdot 10^{-3} \cdot 0.3 \cdot \frac{0.03}{5.024} = 419.9\, N \tag{3}$$

where k is the specific cutting work during transverse sawing of dry pine ($MJ.m^{-3}$), b is the width of the cutting joint (m), and H is the height of the cutting joint (m).

Cutting moment was calculated according to Eq. 4:

$$M_r = F_r.\frac{d}{2} = 419.9 \cdot \frac{0.032}{2} = 6.72\ N \cdot m \tag{4}$$

The resultant stress was calculated according to Eq. 5,

$$\tau = \frac{M_r}{W_k} = \frac{M_r}{\frac{\pi.d^3}{16}} = \frac{6.72}{\frac{\pi \cdot 0.0\ 12^3}{16}} = 19.82\ MPa \tag{5}$$

where W_k is the section modulus in torsion (m^3) and d is the smallest shaft diameter (m).

Mathematical calculations were solved for dry pine. When calculating for other types of wood, it is necessary to consider other parameters such as humidity, time after sharpening the chain, and type of wood. Then, the coefficient k will be given as Eq. 6,

$$k = k_{bor}.K_d.K_w.K_r \tag{6}$$

where k_0 is the basic resistivity for pine during cross cutting ($MJ.\ m^{-3}$), K_d, K_w is the correction coefficients expressing the influence of wood, and K_r is the correction coefficient expressing the effect of blunting.

Table 1. Correction Coefficient Expressing the Effect of Wood (Mikleš 2011)

Wood	K_d	Wood	K_d
Linden tree	0.80	Larch	1.10
Aspen	0.85	Birch	1.20 ÷ 1.30
Spruce	0.9 ÷ 1.0	Beech	1.30 ÷ 1.50
Pine	1.00	Oak	1.50 ÷ 1.60
Alder	1.0 ÷ 1.05	Ash	1.50 ÷ 2.00

Table 2. Correction Coefficient Expressing the Effect of Moisture*

Absolute Moisture	K_w	Absolute Moisture	K_w
8 ÷ 10	0.90	100 ÷ 150	1.10
15 ÷ 20	1.00	150 a viac	1.15
35 ÷ 50	0.95		

*(Mikles 2011)

To reduce the loading moment and the resulting stress, it is necessary to increase the value of the displacement per tooth. The shaft is also stressed for bending. The latter is caused by the tensioning force that is needed to tension the chain. The relationship applies to the tension force F_n, given as Eq. 7,

$$F_n = F_r - F_u \quad N \tag{7}$$

where F_n is the tension force (N), F_r is the cutting force (N), and F_u is the feed force that brings the saw into the cut, (N). For the calculation of the sliding force, the largest calculated cutting force F_r was considered, then Eq. 8 was applied, as follows:

$$F_u = (0.7 \div 1.0) \cdot F_r = 0.7 \cdot 965.77\ =\ 676.039\ \text{N} \tag{8}$$

Tension force is given as Eq. 9:

$$F_n = F_r - F_u = 965.77 - 676.039 = 289.731\ N \tag{9}$$

The bend was solved as a woven beam for the part of the shaft with a reduced diameter at point A. The bending is caused by the tension force acting at the location of the washer and needle bearing in the clutch drum. Because the shaft has different diameters, it was necessary to transform part of the shaft into a uniform diameter.

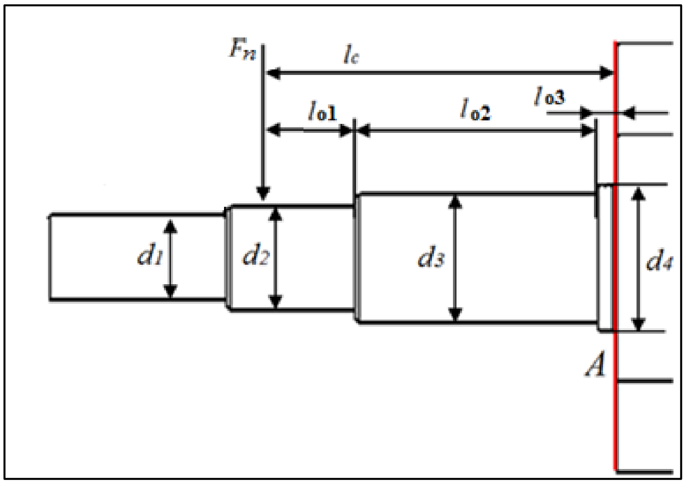

Fig. 6. Loaded part of the shaft

Transformation to uniform mean

The diameter with the largest length d_3 was chosen as the comparative diameter,

$$L_{npi} = l_{oi} \cdot \left(\frac{d}{d_i}\right)^4 \quad \text{mm} \tag{10}$$

where L_{npi} is the transformed length (mm), l_i is the section length (mm), d_i is the section diameter (mm), and d is the comparative average (mm). In Eqs. 11 to 14 there is a recalculation of the transformed length for individual diameters:

$$L_{np1} = l_{o1} \cdot \left(\frac{d}{d_2}\right)^4 = 12.5 \cdot \left(\frac{15}{12}\right)^4 = 24.5 \text{ mm} \tag{11}$$

$$L_{np2} = l_{o2} \cdot \left(\frac{d}{d_3}\right)^4 = 28.5 \cdot \left(\frac{15}{15}\right)^4 = 28.5 \text{ mm} \tag{12}$$

$$L_{np3} = l_{o3} \cdot \left(\frac{d}{d_4}\right)^4 = 2 \cdot \left(\frac{15}{17}\right)^4 = 1.2 \text{ mm} \tag{13}$$

$$L_{np4} = l_{o4} \cdot \left(\frac{d}{d_3}\right)^4 = 7.5 \cdot \left(\frac{15}{15}\right)^4 = 7.5 \text{ mm} \tag{14}$$

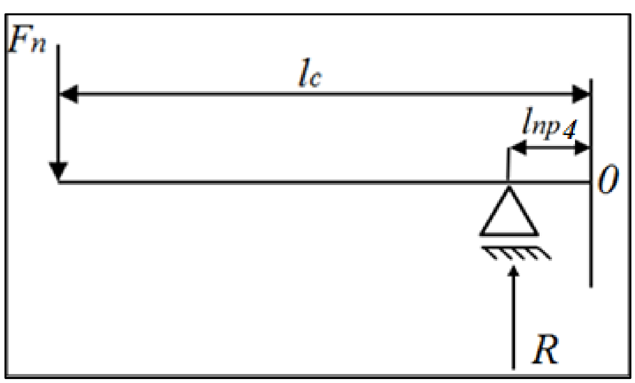

Fig. 7. Simplified shaft diagram

Total length was calculated per Eq. 15,

$$l_c = L_{np1} + L_{np2} + L_{np3} += 24.5 + 28.5 + 1.2 = 54.2 \text{ mm} \quad (15)$$

Bending moment (Eq. 16), will be on the length l_c - l_{np4} due to bearing,

$$M_o = F_n \cdot l_c = 289.731 \cdot 0.0467 = 13.53 \text{ N.m} \quad (16)$$

where F_n is the tension force (N), l_c is the total length of the transformed part of the shaft (m), and M_o is the bending moment (N.m).

Bending stress was calculated by Eq. 17,

$$\sigma_o = \frac{M_o}{W_o} = \frac{M_o}{\frac{\pi \cdot d^3}{32}} = \frac{13.53}{\frac{\pi \cdot 0.015^3}{32}} = 40854918.6 \, Pa = 40.86 \text{ MPa} \quad (17)$$

where σ_o is the bending stress (Pa), W_o is the sectional modulus in bending (m^3), and d is the diameter (m). Tension was calculated as shown in Eqs. 18 and 19,

Reduced tension:

$$\sigma_r = \sqrt{\sigma_o^2 + 3 \cdot \tau^2} = \sqrt{40.86^2 + 3 \cdot 45.58^2} = 88.89 \text{ MPa} \quad (18)$$

Allowed tension:

$$\sigma_{dov} = \frac{R_e}{k} = \frac{245}{1.5} = 163 \text{ MPa} \quad (19)$$

where σ_{dov} is the permissible tension (MPa), k is the safety factor, and R_e is the yield strength (MPa). It applies:

$$\sigma_r \leq \sigma_{dov} \quad (20)$$

All calculated values of the parameters based on the process of transverse sawing with a saw chain necessary for the design of the shaft are listed in table 3.

Table 3. Calculated Values

Parameters	V_c (m.min^{-1})	V_f (m.s^{-1})	F_r (N)	M_r (N.m)	F_n (N)	M_o (N.m)	τ (MPa)	σ_o (MPa)	σ_r (MPa)
Calculated value	301.4	0.03	419.9	6.72	289.7	13.53	19.82	40.86	88.89

RESULTS AND DISCUSSION

The shaft was designed and analyzed in the CREO Parametric program. It is loaded with a torque of 16 N.m in the area of the thread on which the washer will be and a tension force of 290 N. The connections are on the thread through which it is attached to the output shaft of the experimental device. The loading method is shown in Fig. 8. Figure 3 shows the parameters, such as acting forces and the method of constraint entered into the 3D model of the shaft in CREO Parametric, necessary for the analysis.

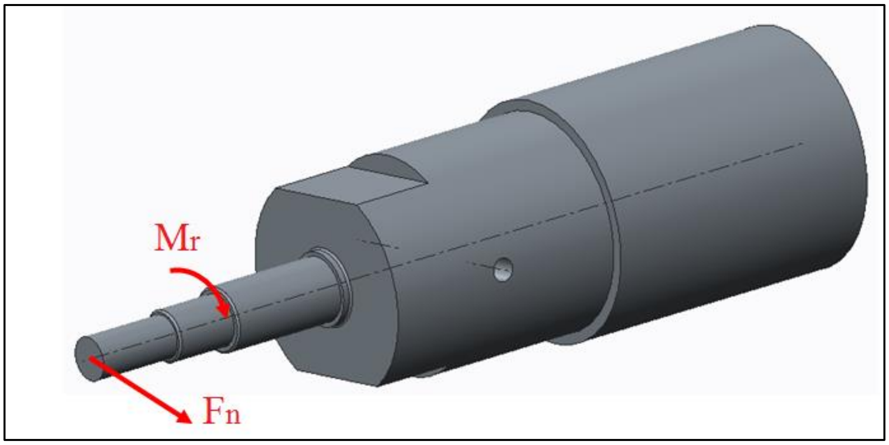

Fig. 8. Shaft load

Finite element method requires dividing the solved area into a finite number of subareas - elements. Therefore, it is necessary to create a mesh of finite elements on the body model (Fig. 10). For each type of element, in addition to dimensions and shape, the number and position of its nodes are characteristic. Nodes of the network are points where we look for unknown parameters of the solution, *e.g.*, displacements and rotations, from which we further calculate stresses. The density and topology of network elements fundamentally affects the quality of the result and the necessary resolution capacity. The advantage of numerical methods is that they allow solving problems on more complex bodies as opposed to analytical approaches, when only elementary bodies can be solved, which occur exceptionally well as machine parts.

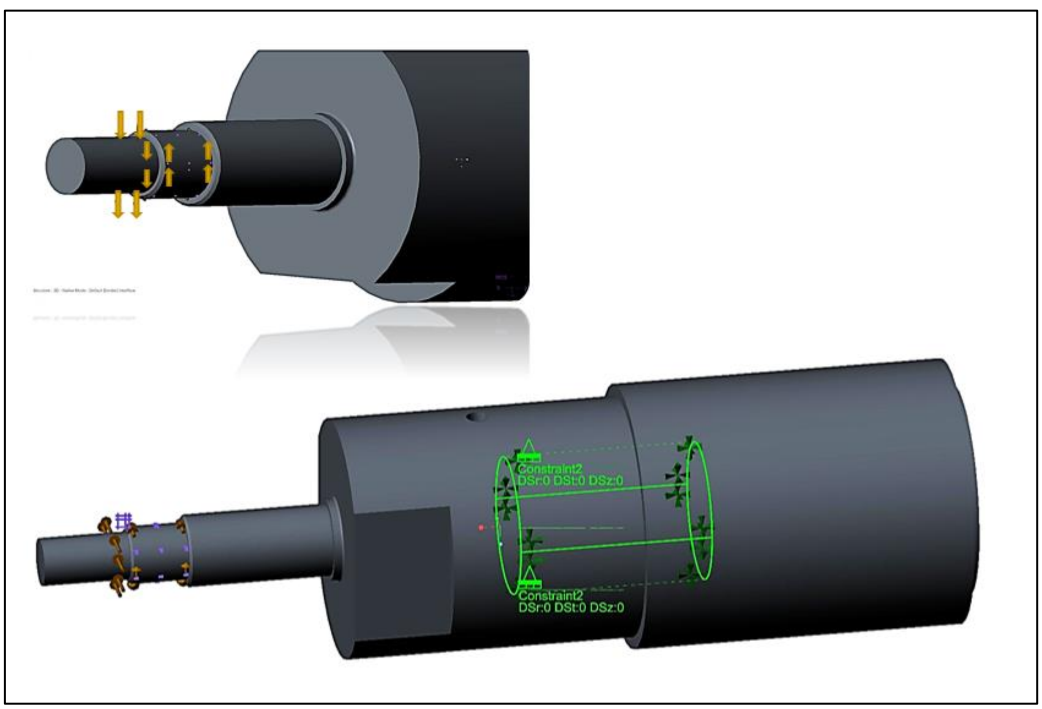

Fig. 9. Shaft attachment

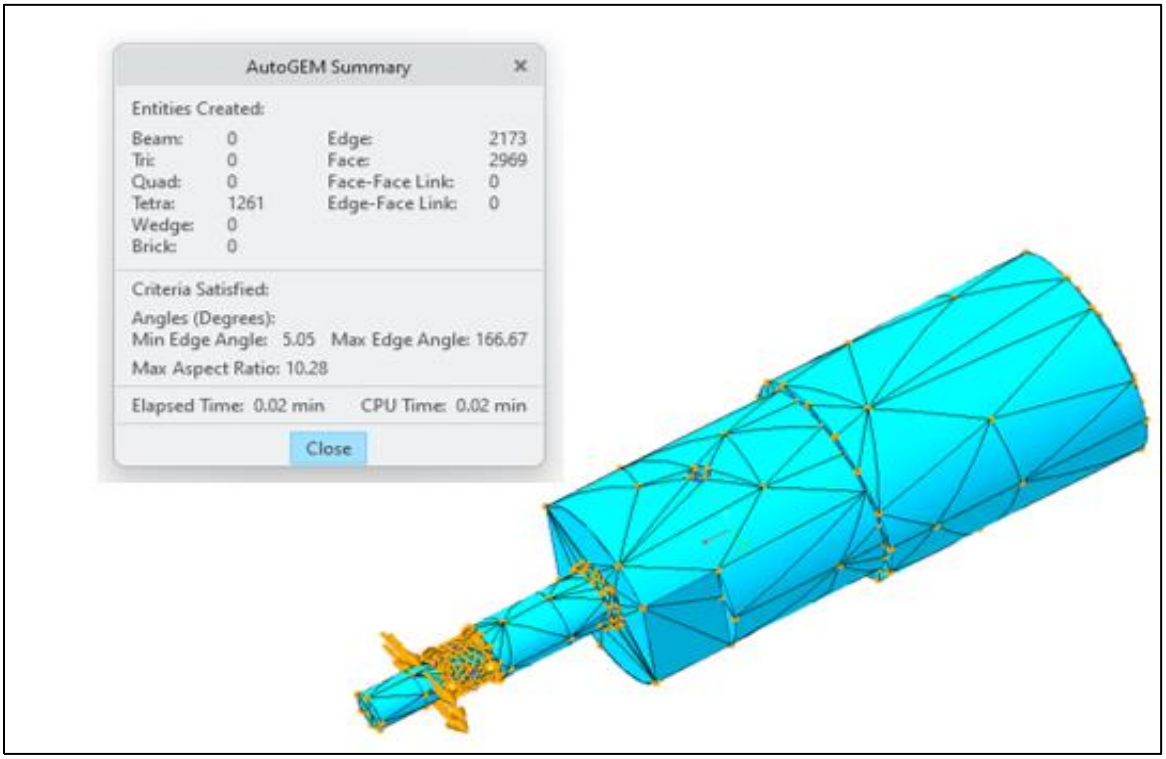

Fig. 10. AutoGem – meshing of finished elements

Figure 11 shows the resulting analysis, which shows where the shaft is most stressed. In this place, the resulting stress is 132.2 MPa. As shown in Fig. 12, the stress is concentrated in the transition from a smaller diameter to a larger one, which is shown in red. Through analysis, higher results were obtained than by calculation. This is attributable to the fact that the model was simplified and there are no roundings, especially in the place of concentrated stress. The stress values do not exceed the permitted value for the shaft material (EN 10025-2 (2004)), also considering the safety factor.

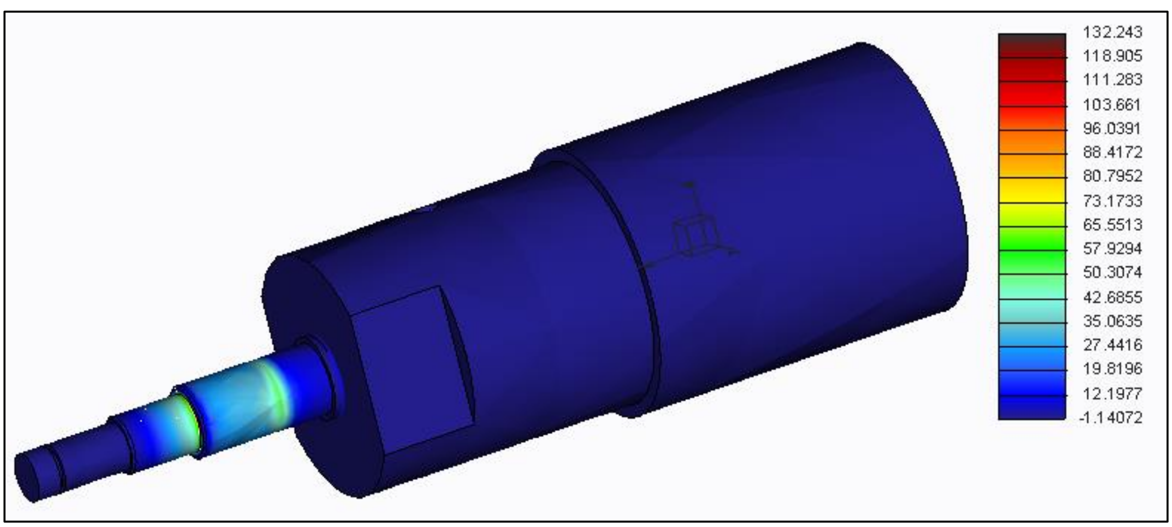

Fig. 11. Shaft stress analysis

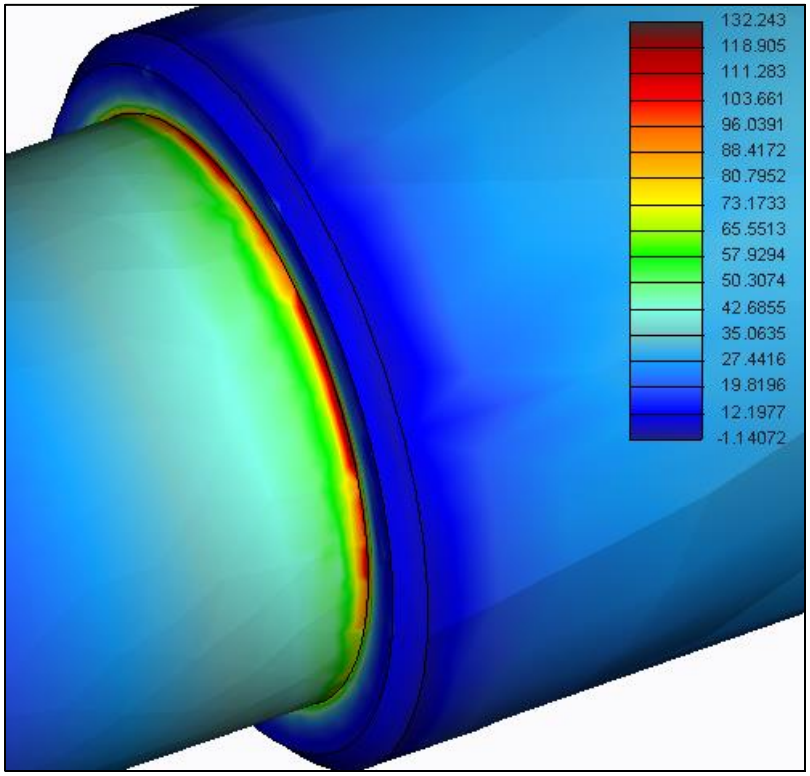

Fig. 12. Detail of the area under the most stress

A number of tools and techniques have been put forth to study the cross-sawing process. In Maciak's (2015) method, the workpiece was moved to the cut using a full chainsaw that was secured to the table. Otto and Parmigiani (2015) employed a test setup that included a saw chain and sprocket on a guide bar. The entire cutting mechanism—which included the engine block, oil pump, clutch drum, sprocket, guide bar, chain, and tensioning device was employed in this investigation and was fixed on the experimental apparatus. Since the solution employed in this study allowed for precise cutting speed setting, it was superior to mounting an entire chainsaw (Maciak *et al.* 2018). The apparatus used in this study is similar to the test apparatus of Otto and Parmigiani (2015), in which only the guide bar with the sprocket was mounted; however, because the entire cutting mechanism was used, it was possible to describe its overall impact on the sawing process.

It was necessary to conduct a strength analysis of the arm's transmission, which is directly impacted by the force from the cutting rectilinear hydraulic motor, in order to determine the stressed portion of the cutting head. Using the finite element method (FEM), the analysis was carried out in Creo simulate. The authors (Mikleš 2012; Hatton *et al.* 2017; Melicherčík *et al.* 2020) have employed this method in their contributions as well.

The experimental device with the cutting mechanism of a chainsaw enables the investigation of cutting conditions during transverse sawing with a saw chain. In practice, the acquired knowledge can bring information for the optimization of mechanisms with a saw chain and settings of suitable parameters in the sawing process, such as cutting speed and feed speed.

CONCLUSIONS

1. Based on the calculation, a cutting force of 419.9 N was found, based on which the cutting moment was 6.72 N.m. Based on the cutting force a and the feed force, the tension force F_n = 289.7 N was found, which loads the shaft for bending.
2. After the transformation of the shaft to a uniform diameter, the resulting reduced stress of 88.89 MPa was calculated, which fulfills the condition $\sigma_r > \sigma_{dov}$ and the designed shaft is suitable for the given device for transverse sawing with a saw chain.
3. The shaft analysis was completed in the CREO Parametric program. The shaft was loaded with the calculated forces and the resulting stress had a value of 132.2 MPa at the point of transition from the smaller diameter to the larger diameter.
4. The parameters mentioned in this work are the basis for further research in the field of chainsaw sawing.

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